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SUMMARY OF COLD-AIR TESTS OF A SINGLE-STAGE TURBINE WITH VARIOUS STATOR COOLING TECHNIQUES

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ABSTRACT

This report summarizes the results of a series of cold-air tests of the aerodynamic effects of coolant on turbine performance conducted at Lewis Research Center. Annular stator exit surveys and single-stage tests were made of a 30-inch diameter turbine using three simulated coolant types of stator blading. The first ejected coolant into the mainstream through a slot in the blade trailing edge. The other two were of the porous skin type, one utilizing a self-supporting shell with discrete holes for variable porosity, and the other a wire mesh skin wrapped around an internal support strut with individual metering orifices and supply passages. Results are presented in terms of stator exit wake traces, stator effeciencies, stage efficiencies, and rotor efficiencies.

INTRODUCTION

To meet their performance requirements, many advanced gas turbine engines must operate at high turbine inlet temperatures. In some cases, these temperatures are high enough that the turbine blading must be cooled to avoid exceeding the stress and oxidation limitations of currently available materials. The general method for cooling the blading is to bleed air from the compressor, direct it through the turbine blading for cooling, and then discharge it from the blade into the main gas stream. The means of discharging the coolant from the blading into the main gas stream are different for different cooled blade designs. In Ref. 1, an anlaysis is presented which indicates that different means of coolant ejection affect turbine performance quite differently. To study these findings experimentally, an investigation is being conducted at the NASA Lewis Research Center to determine the effect on turbine blade row and stage performance of several means of coolant discharge that are typical of different cooled blade designs.

The experimental program on the effect of stator coolant discharge on single-stage performance is complete. A basic turbine having solid stator blades was initially tested (refs. 2 to 4) in order to establish reference performance for comparative purposes. The stator blading was then replaced with three separate blade rows, each having the same profile shape as the solid blades but differing in the manner of simulating coolant ejection. The first ejected coolant into the mainstream through a slot in the blade trailing edge. The other two were of the porous skin type, one utilizing a self-supporting shell with discrete holes for variable porosity, and the other a wire mesh skin wrapped around an internal support strut with individual metering orifices and supply passages. A complete description of the blades tested and the comparative results can be found in Refs. 5 to 11.

The purpose of this report is to summarize the results of this single-stage stator coolant study. A brief description of the different blades tested will be given. The results of annular total pressure surveys made downstream of each stator blade row will then be compared as well as the resulting stator efficiencies. Next, turbine stage efficiencies and calculated rotor efficiencies will be compared from the turbine stage tests. Included is some discussion of the differences between the blading investigated and practical blading for high temperature turbine applications.

DESCRIPTION OF BLADES TESTED

The analysis of Ref. 1 indicates that the effect of the coolant on turbine performance is primarily a function of the amount, location, direction, and energy level of the coolant with respect to the primary air at the point of ejection. In view of this, two extreme cases were selected for study with respect to the location and direction of coolant ejection. A profile sketch of the stator blades studied are shown in Fig. 1. They are the slotted trailing edge blade on the left and two types of porous skins on the right. All three have the same outer shell profile as the solid uncooled blade shown at the top of the figure. The trailing edge slot blades ejected all of the coolant through a slot in the trailing edge in a direction generally the same as the adjacent primary air. The two types of porous skin blades tested, the discrete holes and the wire mesh, ejected air around the entire periphery of the blading and in a direction generally normal to the adjacent primary air.

A photograph of the three cooled type blades tested is shown as Fig. 2. The blade on the left had a slot extending along the entire length of the trailing edge. The discrete hole blade was self supporting and the variation in coolant flow through the blade was controlled by varying the size and spacing of the holes around the surface. The distribution of the coolant around the periphery of the wire mesh blade was controlled by the size of the metering holes in the orifice plate that can be seen at the top of the blade. A photograph of a wire mesh blade in various stages of the fabrication is shown in Fig. 3. The wire mesh blade was made by wrapping and welding a wire mesh around internal struts. The electron beam welds between the mesh and the internal strut can be seen in the figure.

A number of qualifying statements are now made with respect to the tests conducted using these blades. First, the relative performance of the blades were determined by measuring and comparing the total pressure losses at the exit of the blades. It was believed that temperature ratio would have only a small effect on these measurements. Therefore, all tests were conducted with cold air. with no attempt to match any temperature ratio between the coolant and primary air. Also, for simplification, no special provisions were made to simulate internal pressure drops which might be encountered in an actual cooled blade (see fig. 1). For example, the slotted blade shown in Fig. 1 required very little pressure drop to eject the coolant out the trailing edge. On the other hand, the porous skin types in the figure required a considerable pressure drop of the coolant air in order to be ejected through the porous skin. The effect of some of these differences will be discussed later.

For each type of hollow blade tested, a static pressure tube was inserted into the forward cavity near the blade leading edge. These taps were used during tests to set the internal pressure at the forward stagnation position of the blade about 1 inch of Hg higher than turbine inlet total pressure. This insured flow out of the blade around the entire periphery of the porous skin blades and was arbitrarily selected as the design coolant-to-primary air inlet pressure ratio for all of the blades tested. A photograph of the completed annular assembly of the wire mesh stator blading prior to installation into the test facility is shown in Fig. 4. A photograph of a portion of the test facility showing the inlet piping for both the primary and coolant air systems is shown in Fig. 5. The housing for the 30-inch turbine used for the study is at the middle left side of the photograph, with flow from right to left.

STATOR TEST RESULTS

Effect of Coolant Type on Weight-Flow Rate

A comparison of the influence of the manner of ejecting coolant on the total flow rate through the stator is shown in Fig. 6 at design stator pressure ratio. The design pressure ratio was set to produce the required stator outlet hub Mach number of about 0.9. The coolant flow used in this report is shown as a percentage of the primary flow entering the stator prior to coolant ejection. It is apparent from Fig. 6 that the total flow increased for the blades with trailing edge slots and remained essentially constant for both types of porous skin blades. The throat area available to the primary flow for the porous skin blading decreased due to the introduction of the coolant through the skin upstream of the throat. Because all of the coolant for the trailing edge slots was ejected downstream of the throat, the controlling flow area for the primary flow remained essentially unchanged. At a cooling flow rate of 7 percent, the total flow for the slotted blades increased 6.6 percent which indicated that the coolant, in effect, simply filled in the space downstream of the trailing edge slot.

Stator Wake Traces

Surveys were made about 0.25 inch axially downstream of each stator configuration using a total pressure probe set at the average free-stream flow angle. The surveys were made for coolant flow rates up to 7 percent for design overall stator pressure ratio. A comparison of the mean-section stator wake traces is shown in Fig. 7 for the 3 different blades tested. The wake traces are shown as a drop in total pressure as a percent of inlet total pressure for coolant flows up to 7 percent. The deficit in area shown by the wake for positive values of pressure drop is an indication of blade losses. The blip for 7 percent coolant for the slotted blade (negative value of pressure drop) means that the total pressure of the ejected coolant was higher than the inlet total pressure of the primary air. This is because the coolant flow rate was varied by changing the cavity pressure of the coolant within the blade.

There is a 0 percent coolant wake trace shown for each blade tested. Of particular interest is a comparison of what happens to the width of this wake as coolant is added. As can be noted for the trailing edge slot, the coolant filled in the wake from the trailing edge of the blade. However, its effect did not extend outward into the region of the primary flow. This was not the case for the porous skin types. As flow was added, there was a large build up of low energy fluid on the suction side of the blading that extended the loss region considerably out into the free stream area.

Stator Efficiencies

The results of the full annular total pressure surveys were used, with other data, to calculate and compare stator efficiencies. Two efficiencies were calculated for each blade tested. The primary air efficiency η_p relates the kinetic energies of both the primary and coolant flows to the ideal kinetic energy of the primary flow only. In equation form, the stator primary efficiency is expressed as

$$\eta_{p,\,stator} = \frac{w_p V_{p,1}^2 + w_c V_{c,1}^2}{w_p V_{p,id,1}^2} \tag{1}$$

where station 1 is at the stator outlet. The thermodynamic stator efficiency η_t relates the kinetic energies of both flows to the ideal energies of both flows. In equation form

$$\eta_{t,\,stator} = \frac{w_p V_{p,1}^2 + w_c V_{c,1}^2}{w_p V_{p,id,1}^2 + w_c V_{c,id,1}^2} \tag{2}$$

It is felt that the primary efficiency is more relevant to this type of study for two reasons. First, it is a better indication of the perturbing effect that the coolant has on the performance of the primary flow without coolant. This advantage appears to make primary efficiency more adaptable to gross engine studies than thermodynamic efficiency. Secondly, the selection of a meaningful definition of the ideal coolant energy in the denominator of the thermodynamic efficiency (eq. (2)) is difficult when the particular blades and tests conducted are considered. Therefore, particular emphasis will be placed on primary efficiencies, although thermodynamic efficiencies will be shown for the stators for general interest.

A comparison of the turbine stator primary efficiencies of the blades tested is shown in Fig. 8. A circle is shown on each curve to compare the amount of coolant flows ejected for values of coolant-to-primary air inlet pressure ratio of unity, which is close to the design value of slightly over unity referred to earlier. As expected, more flow was ejected from the slotted trailing edge than either of the porous skins due to less internal pressure drop, as discussed for Fig. 1. At this condition, the coolant flow rates were about 5, 3, and 2 percent for the slotted, discrete holes, and wire mesh blades respectively. The stator primary efficiencies for both porous blades remained nearly constant at about 96 percent over the 7 percent range of coolants tested. The efficiency values at coolant rates less than those indicated by the circles on Fig. 8 have little significance for the porous skin blading because the pressures inside the blades were lower than primary air inlet pressure. For these conditions, inspection of Fig. 1 indicates that primary inlet air would "leak through" the porous skin blades from the pressure to the suction side of the blade and probably result in a larger buildup of low energy fluid on the suction side of the blade with attendant higher losses. This was especially true for the discrete hole blade because it was not compartmented internally as was the wire mesh blade. The larger leakage at 0 coolant flow for the discrete hole blading is evidenced by the larger mean section wake traces of Fig. 7 and the lower efficiencies at 0 coolant flow in Fig. 8.

The fact that the stator primary efficiencies of the porous skin blading remained essentially constant indicates that the ejected coolant had very little, if any, kinetic energy at the exit of the blade row. Also, the location and direction of the coolant ejection could have adversely affected the performance of the primary air, such as possible flow separation from the suction surface as will be discussed later.

In contrast, the stator primary efficiency of the slotted trailing edge blades continually increased as coolant was added. Fig. 8 shows that the efficiency of the slotted blades increased from about 97 percent at zero coolant flow to about 106 percent at 7 percent coolant flow. The 9 percent increase in efficiency is larger than the increase in coolant flow. Eq. 1 shows that this can occur when the velocity of the coolant is higher than the velocity of the primary air ($V_{c,1} > V_{p,1}$). As discussed for the wake traces of Fig. 7, this was true for the slotted blading at the high coolant flows when the coolant inlet pressure (and ideal energy) of the coolant was higher than that of the primary air. When the inlet pressures were equal (circles in fig. 8) the coolant flow rate was about 5 percent and the efficiency of the slotted blade was 101.7 percent, or a 4.7 percent increase over that with 0 percent coolant flow. This indicates that the kinetic energies of the coolant and primary air were about equal at the blade exit $(V_{c,1} \cong V_{p,1})$.

A comparison of the stator thermodynamic efficiencies for the blades tested is made in Fig. 9. The selected ideal coolant energy charged in the denominator of Eq. (2) was arbitrarily based on the energy between the inlet total pressure of the coolant and the static pressure at the exit of the blade row. The ideal energies of the coolant and primary air for each blade type tested are equal at coolant flows indicated by the circles of Fig. 9. There is an obvious difference in the level of efficiencies experienced by both porous skin blades when compared to the blades with trailing edge slots. The thermodynamic efficiency of the slotted blades varied very little from its value at 0 coolant flow and is close to the value of the solid blade efficiency of about 97 percent. The porous skin types, on the other hand, continually decreased in efficiency as coolant was added. As an example, at 7 percent coolant flow, the efficiency of the porous skin blades decreased about 11 points.

The reason that the efficiency divergence between the slotted and porous blades is greater for the thermodynamic efficiencies (about 11 points from fig. 9) than for the primary efficiencies (about 9 points from fig. 8) is due to higher required pressure drops for the coolant to be ejected through the porous skins. This, then, effects the relative ideal coolant velocities in the denominator of Eq. 2.

As mentioned previously, the blades tested were simplified in construction and comparative observations for use in actual hot turbine designs should be tempered with this in mind.

STAGE TEST RESULTS

Turbine Stage Efficiencies

A photograph of the solid rotor used to determine stage performance of the solid stators and the three hollow stator configurations is shown in Fig. 10. The tip diameter of the rotor is 30 inches and the blades are 4 inches long. The indicated blade profiles, including the leading and trailing edges, are thick, simulating the profile that might be considered for high temperature turbines. Cold-air performance maps were made for each turbine over a range of speed and pressure ratio that bracketed the design equivalent work output of the base solid turbine of 17 Btu per pound. The turbine stage efficiency at this design condition was reported in Ref. 4 to be 0.923. Fig. 11 shows a comparison of the stage primary efficiencies of the various turbines for coolant flow rates up to 7 percent. All test results shown are for each turbine at design speed and design primary work output of 17 Btu per pound. The primary work output is equal to the turbine power output divided by the primary weight flow. In terms of measured quantities it is equal to the numerator of the expression used for turbine stage primary efficiency

$$\eta_{p,\,stage} = \frac{\Delta h_p}{\Delta h_{p,\,id}} = \frac{\dfrac{\pi N \tau}{w_p 30J}}{\Delta h_{p,\,id}} \tag{3}$$

As expected, the trends of stage primary efficiencies shown in Fig. 11 are quite similar to those for stator primary efficiencies discussed in Fig. 8. However, an interesting comparison is made at 0 coolant flow rate between the two figures. Both figures show that the slotted stator blades resulted in efficiencies about the same as those for the solid stator blades. The stator efficiencies of the porous skin blades were about 1 point lower (fig. 8). If the rotor performed the same for all turbines, then the stage efficiency with the porous stator blades should be no more than 1 point smaller than that with the slotted blade. However, Fig. 11 shows this difference to be about 2 points. This suggested that the performance of the rotor was poorer when tested behind the porous skin stators than behind the slotted or solid stators.

Rotor Efficiencies

One reason that the rotor might differ in performance could be the difference in circumferential flow patterns entering the rotor for the various stator types. The considerably thicker wakes from the porous stators (fig. 7) could cause an additional entrance loss for the rotor blades. If this is true, then one might expect the amount of coolant ejected from the stators to also influence rotor efficiency. Consequently, rotor efficiencies were calculated using experimental data obtained from stator survey tests and stage tests for all of the turbines. The equation used (ref. 9) for rotor efficiency was

$$\eta_r = \frac{\Delta h_r}{\Delta h_{r,id}} = \frac{\dfrac{\tau N \pi}{(w_p + w_c)30J}}{\dfrac{V_1^2}{2gJ} + \left(h_1 - h'_{2,id}\right)} \tag{4}$$

which expresses the work of the total flow through the rotor to the ideal rotor work. The results are shown in Fig. 12. The calculated rotor efficiency of the slotted blade turbine was highest at 0 coolant flow and was essentially unaffected over the range of coolant flows tested. At 0 coolant flow the porous blades resulted in calculated rotor efficiencies about 1 point lower. Also, this loss increased with added coolant. At 7 percent coolant flow rate, there was an additional rotor loss of about 1 point. These results tend to confirm the fact that large circumferential gradients in flow conditions can adversely affect the performance of a following blade row. An additional influencing factor could also be the axial spacing between such blade rows. The axial clearance between stator and rotor blades for all tests reported herein was about 0.5 inches. The rotor was moved back axially an additional 1 inch behind the wire mesh porous skin stator assembly and retested. Preliminary results indicate that some of this additional loss within the rotor was recovered by allowing the flow leaving the stator blades to more thoroughly mix before entering the rotor.

PARTIALLY BLOCKED POROUS SKIN

The last portion of this report is concerned with the fact that the stator blades tested were not designed for actual hot-turbine applications. As discussed earlier, the various blades used for the general tests described were purposely selected to represent two extremes in terms of location and direction of coolant ejection. From the sketch of the blades tested, Fig. 1, it is recognized that there can be a very large number of various cooling schemes or combinations of schemes that could fall somewhere between these two extremes. For example, the trailing edge slot blade tested required a relatively small pressure drop to eject the coolant. For an actual blade, of course, the particular internal flow passages would decrease the energy level of the coolant as it is ejected from the slot. Also, practical considerations should be made for porous skin blades. For example, it may be desirable from either an aerodynamic or heat transfer position to selectively block off various portions of the blade. To demonstrate this, the wire mesh blades were retested with the back 2/3 portion of the suction surface and the trailing edge region sealed (see fig. 13). All of the coolant, then, was ejected from the forward 1/3 of the suction surface and the major length of the pressure surface. Annular total pressure surveys were made at the stator exit in the same manner as those used to obtain similar results previously shown for the other turbines.

A comparison of the mean section total pressure wake traces for this test with the results from the full ejection blade is made in Fig. 14. A considerable difference in the pattern of the wake traces can be noted. For the full ejection case on the left, there was a large build up of low energy fluid on the suction side referred to earlier. For the partially blocked blade on the right, the thickness of the low energy fluid on the suction surface became substantially less. And, because a larger portion of the air was ejected out the pressure surface, the thickness of the low energy fluid on the pressure side of the blade was larger. It appears that if the general width and depth of the two boundary layers are compared, the partially blocked blade on the right should have a higher performance. This is shown to be true in Fig. 15, where a comparison of the stator primary air efficiencies is made for both the partially blocked and full ejection wire mesh blades over the 7 percent range of coolant flow. A considerable increase in stator kinetic energy output resulted from selective ejection through the porous blade. Whereas the primary efficiency of the full ejection blade varied only a small amount from its value at 0 coolant flow, there was a 5 point increase in efficiency for the partially blocked blade at a coolant flow rate of 6 percent. In other words, because a large portion of the coolant was ejected near the front of the partially blocked blade, it expanded to a kinetic energy level at the exit of the blade nearly equal to that of the primary air (eq. (1)). Probably the most significant difference is in the flow conditions and loss buildup on the back portion of the suction surface. Ref. 8 indicated that the losses were quite high in this diffusing region due to the required rise in static pressure from the point of injection to the exit of the blade and possible flow separation from the blade surface. It appears, then, that this is one region of the blade where ejection of coolant normal to the primary gas stream is highly undesirable. Whether or not redirecting coolant ejection in this area in the direction of the primary flow would improve conditions probably depends on the boundary layer history prior to ejection, the energy level of the coolant, and the local diffusion characteristics from this point to the blade exit.

CONCLUDING REMARKS

In summary, the tests described indicate that the aerodynamic effect of coolant on turbine performance is a function of the amount, location, direction, and energy level of the coolant with respect to the primary air at the point of ejection. The two extreme cases selected for study resulted in considerable differences in turbine stator and stage performance. In addition, large circumferential gradients in wake patterns from the porous stators caused an additional loss within the rotor that was accentuated with increased coolant flow rates. The performance of the wire mesh porous stator improved when the diffusing portion of the suction surface was blocked. Although the blades tested were not designed for actual cooling, the results indicate that the aerodynamic penalty can be significant and should be considered when evaluating a given cooling application.

NOMENCLATURE

g	force-mass conversional constant, 32.174 ft/sec^2
h	specific enthalpy, Btu/lb; J/kg
Δh	specific work output, Btu/lb; J/kg
J	mechanical equivalent of heat, 778.16 ft-lb/Btu
N	rotational speed, rpm
p	absolute pressure, lb/ft^2; N/m^2
V	absolute gas velocity, ft/sec; m/sec
w	mass-flow rate, lb/sec; kg/sec
η	efficiency
τ	torque, ft-lb; N-m

Subscripts:

c	coolant
id	ideal
p	primary
r	rotor
t	thermodynamic
0	turbine inlet

1 stator outlet, also rotor inlet

2 rotor outlet

Superscript:

' total state

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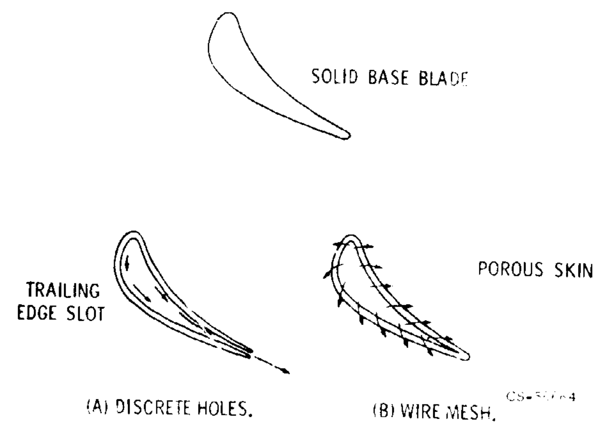

Figure 1. - Cooled turbine blade types.

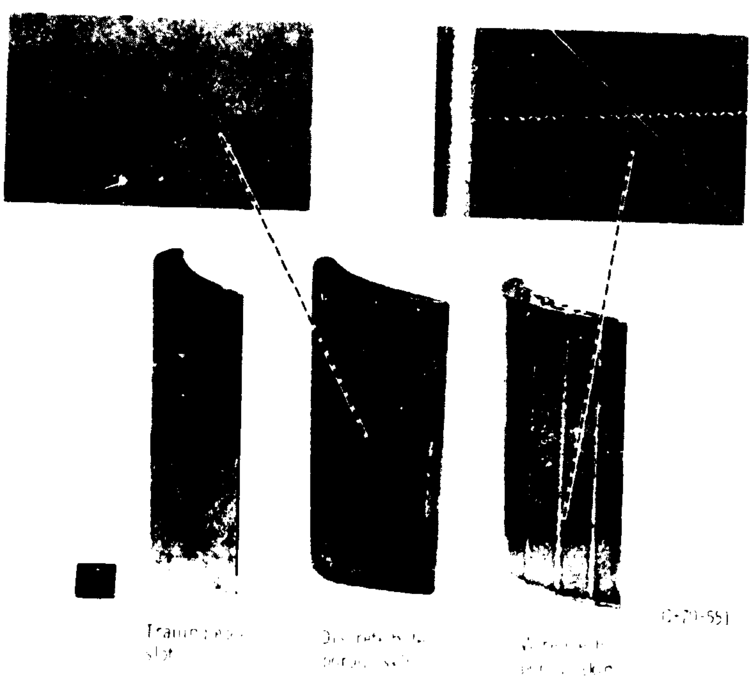

Figure 2. - Stator blades tested.

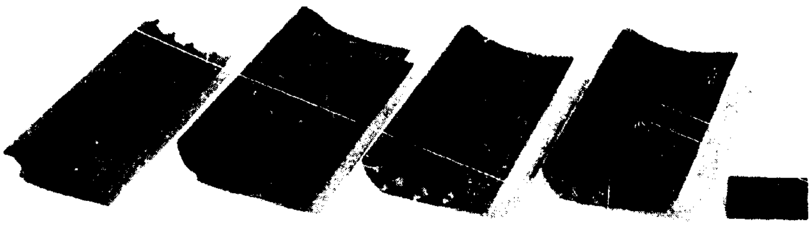

Figure 3. - Assembly of wire mesh blade.

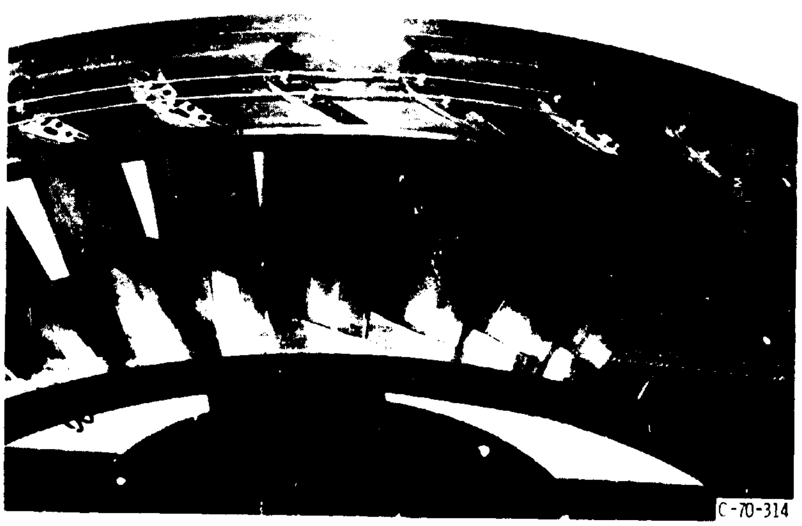

Figure 4. - Assembled wire mesh stators.

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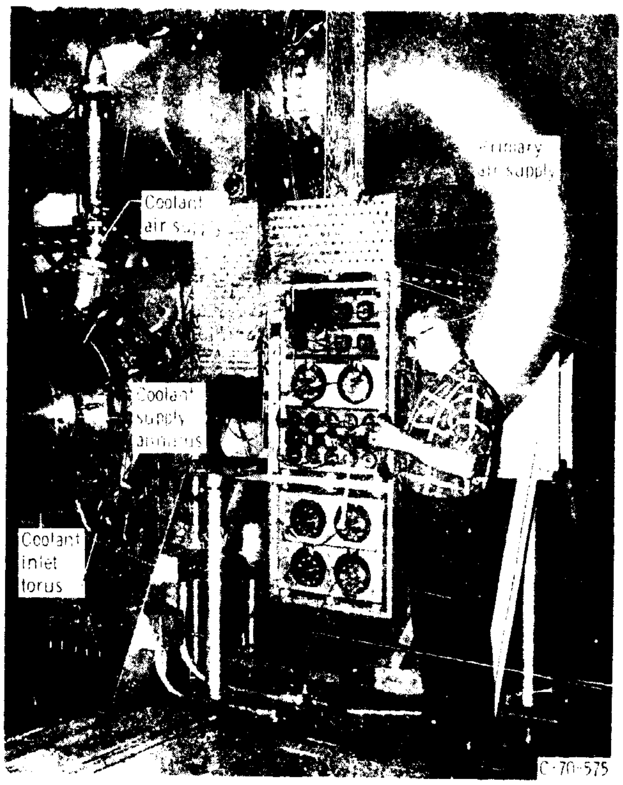

Figure 5. - Test facility.

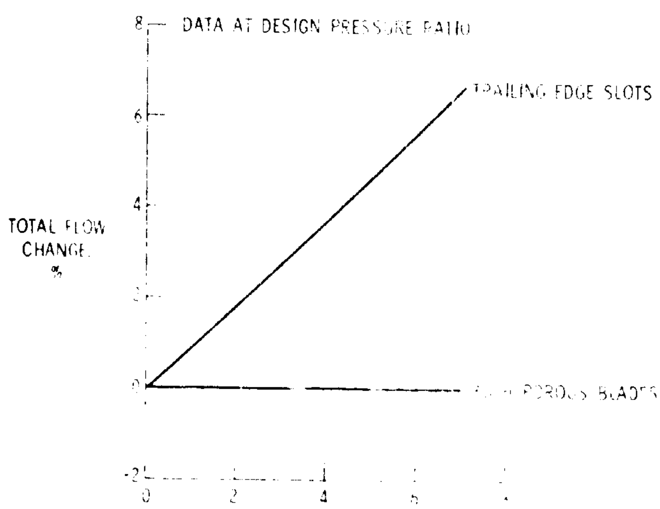

Figure [illegible] - Effect of coolant [illegible]

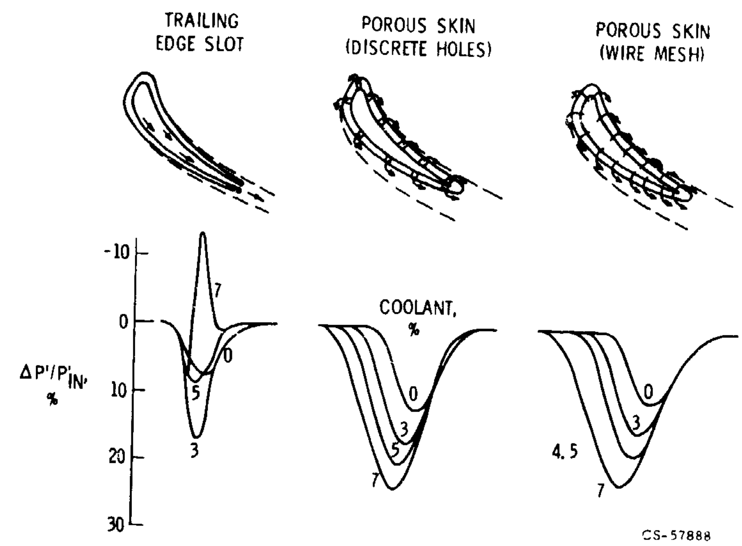

Figure 7. - Stator wake traces.

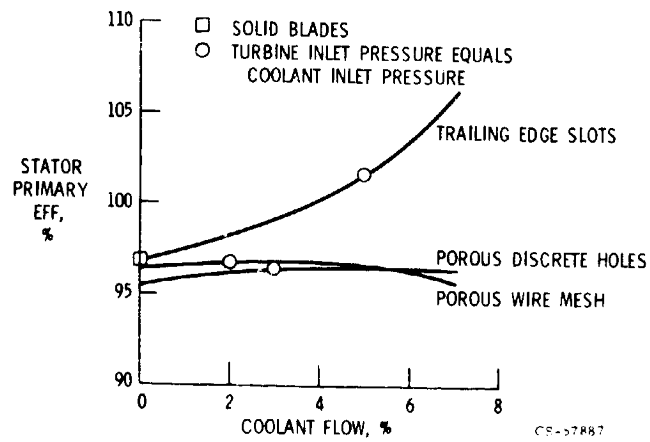

Figure 8. - Turbine stator primary efficiencies.

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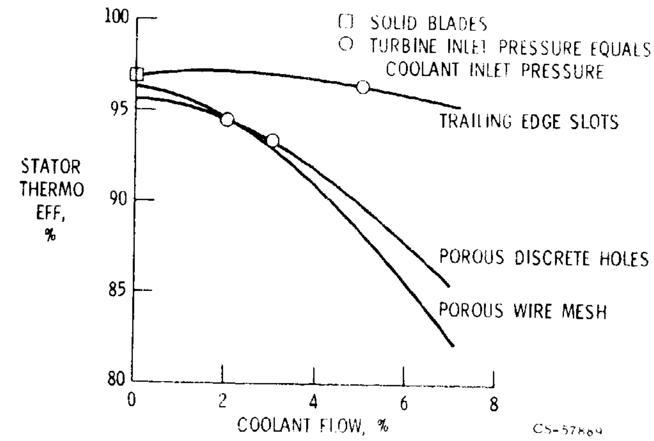

Figure 9. - Turbine stator thermodynamic efficiencies.

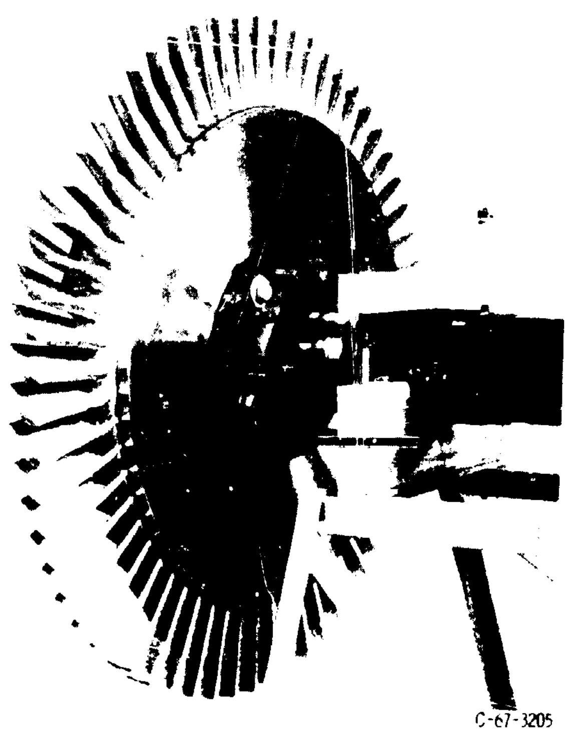

Figure 10. - Turbine rotor assembly.

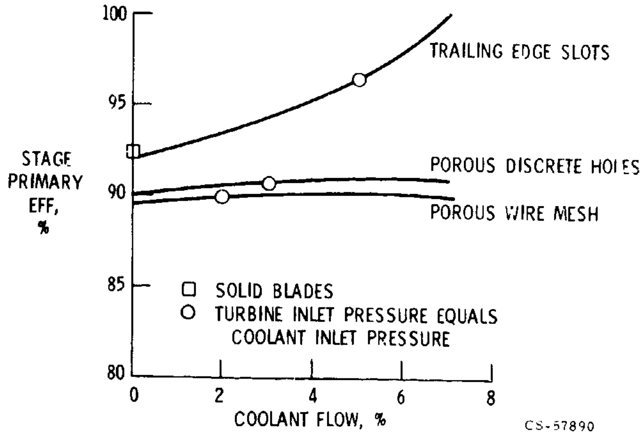

Figure 11. - Turbine stage primary efficiencies.

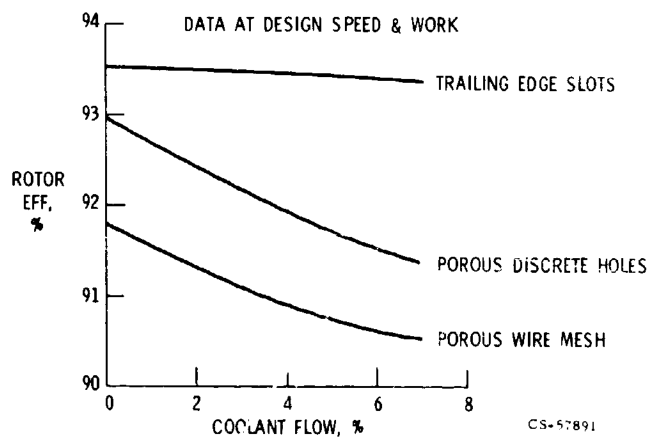

Figure 12. - Effect of flow pattern on rotor efficiencies.

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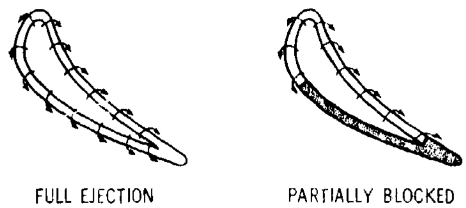

Figure 13. - Partial blockage of wire mesh blades.

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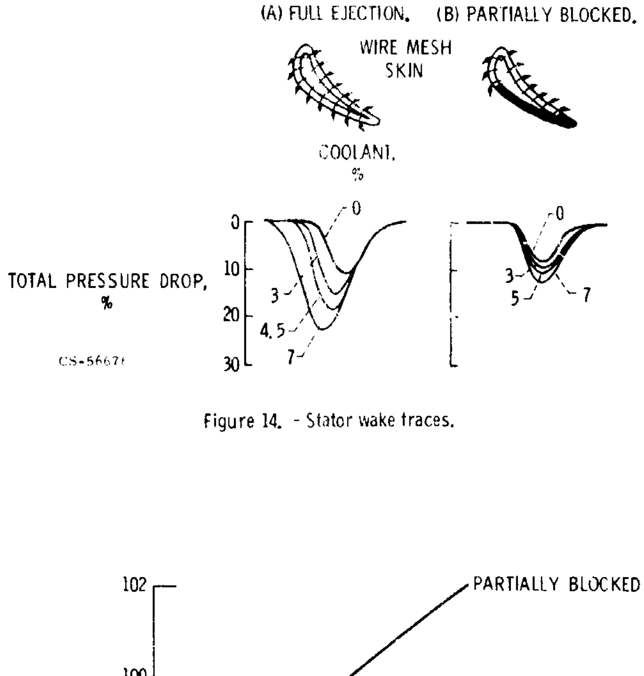

Figure 14. - Stator wake traces.

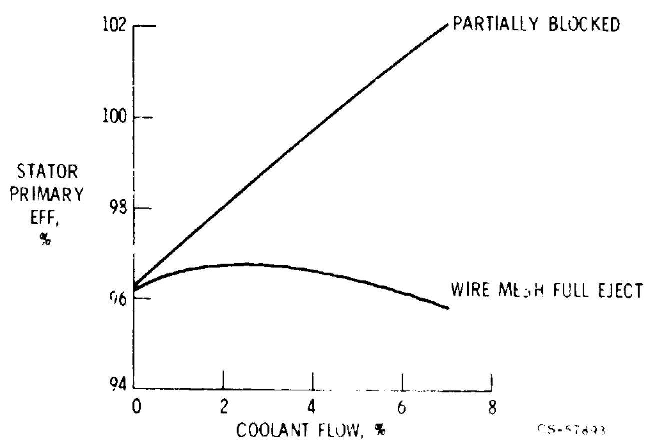

Figure 15. - Partially blocked wire mesh stator blades.